

Investigation on a Stirling refrigerator with thermal buffer tube driven by an oil-lubricated compressor

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Abstract

In this article, a special configuration of Stirling refrigerator for domestic refrigeration purpose is introduced. A thermal buffer tube is installed between the refrigerator cold-end and the expansion piston to improve the system reliability by moving the expansion piston from low temperature to ambient temperature. Furthermore, a commercial oil-lubricated dual-piston compressor is modified to drive the refrigerator, inside which an elastic membrane is used to transfer acoustic work and separate the working gas of the refrigerator from that of the compressor. Experimental investigations on the refrigerator are performed using helium as the working fluid and a cooling power of 200 W at $-78\text{ }^{\circ}\text{C}$ is achieved at 15 Hz working frequency and 2.5 MPa mean pressure. Meanwhile, a rough estimation of the refrigerator COP in terms of cooling power divided by input acoustic power gives the value of 0.64. It gives the possibility of building a low-cost, high efficiency domestic refrigerator.

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Keywords: Domestic refrigerator; Design; Compression system; Stirling; Pulse tube; Membrane; Experiment

Etude sur un réfrigérateur Stirling muni d'un tube servant de tampon thermique et d'un compresseur lubrifié à l'huile

Mots clés : Réfrigérateur domestique ; Conception ; Système à compression ; Stirling ; Tube à pulsation ; Membrane ; Expérimentation

1. Introduction

In the past two decades, growing concerns over environmental issues caused by the working medium in conventional refrigerators have led to increased research activities

in search of substitution technologies for vapor-compression cycles. Stirling refrigerator is one of the alternative technologies with chemically inert gases, such as helium, as its working fluids, which is completely environment-friendly. Furthermore, the Stirling refrigerator is always described as an ideal machine, which can theoretically achieve the Carnot efficiency.

In 1834, the reversed Stirling cycle was firstly proposed as a refrigerator by Alexander Kirk for ice making. After about one century's effort, the first commercial

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Nomenclature

t	time (s)
U_{comp}	volume velocity of compression piston ($\text{m}^3 \text{s}^{-1}$)
U_{expa}	volume velocity of expansion piston ($\text{m}^3 \text{s}^{-1}$)
V_s	piston swept volume (m^3)
x	axial distance (m)

Greek symbols

θ_0	initial phase angle
ω	angular frequency (rad s^{-1})

two-piston type Stirling refrigerator was produced by Philips in 1950s [1]. However, due to several reliability problems, this two-piston type Stirling refrigerator developed slowly especially in the area of domestic refrigeration. Firstly, the expansion piston always works in a low temperature environment, which suffers a big temperature gradient and reduces its reliability. Secondly, the lubricating oil used for moving parts of the compressor, before invention of the oil-free pressure wave generators, can contaminate the working gas of the refrigerator and leads to a requirement of the oil removal devices or in-time maintenances, which eventually increases system complexity and reduces its reliability.

The thermal buffer tube (TBT) placed between the cold-end and the expansion piston could solve the first problem, with which the expansion piston is moved from low temperature environment to ambient location and greatly improves the reliability. So far, this modified String cryocooler, so-called free warm expander pulse tube cooler (FWEPTC), is mainly for cryogenic applications [2–4]. Here, our work is focused on the refrigerator working within the temperature range of two-stage vapor-compression refrigerators, which is similar to FWEPTC in configuration but has several hundred watts cooling capacity at a much higher temperature and thus requires the recovery of expansion work. As to the second problem, it can be solved by using the oil-free pressure wave generators, such as linear compressors [5]. Nevertheless, high-power oil-free compressors required by large cooling capacity refrigerators are still expensive and not easily available. Actually, the commercial oil-lubricated compressors for domestic refrigeration equipment are relatively cheap and very reliable. To make use of these advantages, we have proposed to use an elastic membrane to modify the oil-lubricated compressor, which will be introduced later in detail.

In the following, experimental setup of the oil-lubricated compressor driven TBT type Stirling refrigerator (TBTSR) is given firstly. Then numerical simulation approach is briefly introduced. After that, experimental results are given, including typical experimental pressure waveforms, cool-down curve, influence of working frequency and mean pressure. Next, discussions on working duration of the membrane and the COP estimation of the refrigerator are performed. Lastly, some conclusions are made.

2. Experimental configuration

Fig. 1 gives the diagram of the experimental setup of the TBTSR driven by an oil-lubricated compressor. The TBTSR surrounded by dashed lines consists of five components, including main ambient heat exchanger (HX), regenerator, cold-end HX, TBT and secondary ambient HX, which are successively numbered from ① to ⑤. All HXs are plate-fin type and made of copper. And calculated heat exchange area of main ambient HX, cold-end HX and secondary ambient HX are 0.109, 0.112 and 0.013 m^2 , respectively. Four heating rods are embedded in the cold-end HX to measure cooling power. The regenerator is filled with 150 mesh

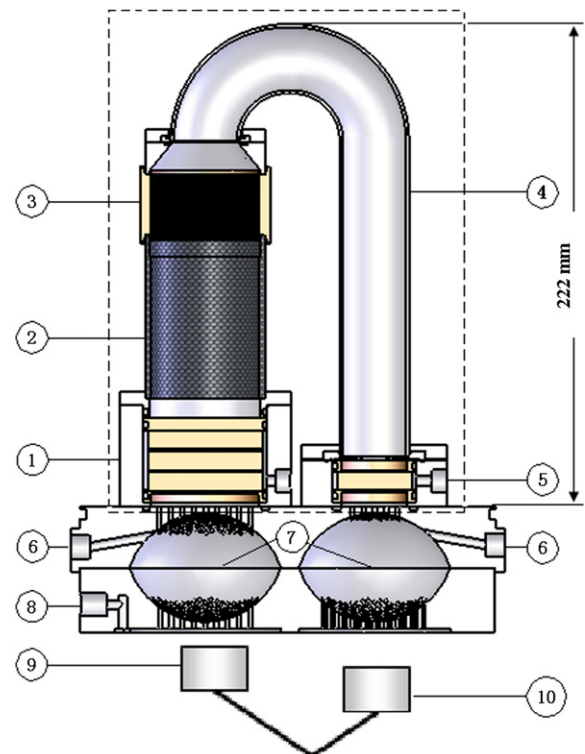


Fig. 1. Diagram of TBTSR driven by an oil-lubricated dual-piston compressor. ① main ambient HX; ② regenerator; ③ cold-end HX; ④ TBT; ⑤ secondary ambient HX; ⑥ pressure sensor interface; ⑦ membrane; ⑧ oil channel; ⑨ compression piston; and ⑩ expansion piston.

stainless steel screens. The main ambient HX, regenerator and cold-end HX are 35, 82 and 30 mm long, respectively, and housed in a 50 mm inner diameter (i.d.) stainless steel pipe. The TBT and secondary ambient HX are 290 and 10 mm long and housed in a 29 mm i.d. pipe. The TBTSR itself is 222 mm high as shown in Fig. 1. All structure parameters are carefully chosen according to numerical calculations to be introduced in Section 3. The refrigerator is housed in vacuum chamber in the experiments.

From ㉔ to ㉔, it gives a novel design of a pressure wave generator consisting of an elastic membrane ㉔ with a protection configuration and an oil-lubricated two-piston compressor represented by two pistons in the figure. The elastic membrane, which is placed in the middle of the two ellipsoid shaped cavities of the protection structure with many 2 mm i.d. flow channels at the top and bottom of each cavity, can not only transfer acoustic wave, but also efficiently separate the working gas in the refrigerator from the contaminated gas in the compressor. Longevity of the membrane strongly depends on its deforming area, that is, larger deforming area will give longer lifetime with a set piston swept volume since it leads to smaller displacement of the membrane. However, large area could bring large void volume of the system resulting in performance reduction of the refrigerator. In our experiment, the membrane is 70 mm diameter comparing to 53 mm diameter of the pistons. The compression piston ㉔ and the expansion piston ㉔ of the compressor are connected with a crank shaft, which can control phase difference between two pistons and allow the expansion work to be recovered. In the experiments, the compression piston leads the expansion piston by 120° , and the swept volumes are both 88 cm^3 . Pressure sensor interfaces ㉔ are used for pressure monitoring. Additionally, the oil channel ㉔ is used for the lubricating oil accumulated before the pistons to flow back to the crankcase. Additionally, an electric inverter is used for changing working frequency and showing the input electrical power of the compressor.

3. Numerical approach

Numerical simulation of the TBTSR is performed for system design and performance improvement. The approach is based on the linear thermoacoustic theory, which is established by Rott [6] and developed by Swift [7] and Xiao [8]. And more simulation details can be found in Refs. [9,10]. The computational input conditions of TBTSR are volume velocities of two pistons, which can be obtained as follow

$$U_{\text{comp}} = \frac{\omega V_s}{2} \sin(\omega t + \theta_0) \quad (1)$$

$$U_{\text{expa}} = \frac{\omega V_s}{2} \sin(\omega t + \theta_0 - 120^\circ) \quad (2)$$

where U_{comp} and U_{expa} represent the volume velocities of compression piston and expansion piston, respectively.

V_s is the piston swept volume, ω is angular frequency, and θ_0 is initial phase angle. The ambient temperature is 27°C and the cooling water temperature is 17°C .

In the simulation, temperature distribution and power flows are most concerned, from which power conversion in the refrigerator can be demonstrated clearly. Fig. 2 gives a typical computation results of axial distributions of temperature, acoustic work flow, total power flow and heat flow, respectively, with cold-end temperature -80°C under working frequency 15 Hz and mean pressure 2.5 MPa; $x = 0$ and $x = 547$ locate at the inlets of two ellipsoid shape cavities. As Fig. 2 shows, thermoacoustic effect happens in the regenerator between working gas and stainless steel screens stacked in the regenerator. Eventually, acoustic work is consumed to pump the heat from cold-end HX to main ambient HX. From plot of the acoustic work flow, the expansion work reaches 57% of the compression work and should be recovered for high efficiency purpose.

4. Experimental results

4.1. Characteristic of pressure waveforms

In the first step of the experiment, efforts have been made to try to obtain proper pressure waveforms, which is important for the operation of TBTSR. In fact, during the cool-down process, equilibrium position of the membrane would move upward in Fig. 1 because the mean pressure decreases in the refrigerator side as the temperature goes down and the mean pressure increases in the compressor crankcase due to dissipation inside. Therefore, the membrane could possibly touch the upper wall of the cavity during the operation and the pressure waveform will not be sinusoidal. To solve this problem, very thin pipes are used to connect both the TBTSR and the compressor crankcase with relatively large

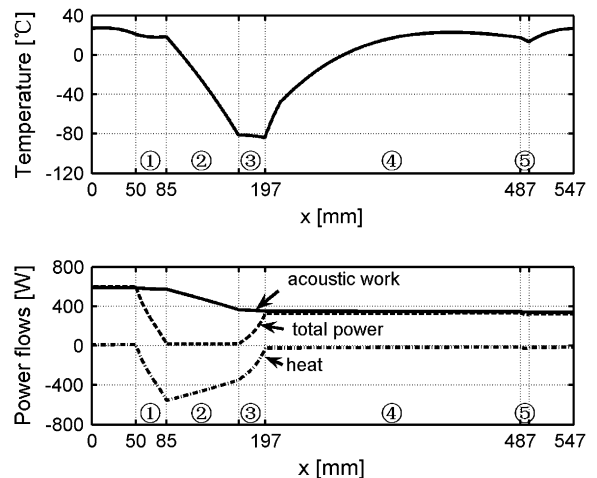


Fig. 2. Computational distributions of power flows and temperature along axial direction.

gas reservoirs, respectively, which can make up for the pressure change in the PTSR and the compressor. Fig. 3 gives typical pressure waveforms at the upper front of the two pistons with helium as the working gas, 2.5 MPa mean pressure and 15 Hz working frequency. It can be found that the two pressure waves were almost the same which indicates small pressure drop loss across the regenerator.

4.2. Cool-down curve and cooling power measurements

Fig. 4 gives the cool-down curve with cooling capacity measuring process with helium under 15 Hz frequency and 2.5 MPa mean pressure. At the beginning, the cold-end temperature equaled the ambient temperature. After the system run, the cold-end temperature reached -100°C in about 4 min, then 100 W heat load was given to the cold-end and the temperature rose to -105°C after 20 min. Successively, different cooling capacities were obtained under different cold-end temperatures. To accelerate the experimental process, larger heat powers than expected were given first when the temperature went up from lower to a higher one, then the heat power was reduced to the expected value, which resulted in the small peaks before every temperature stage.

4.3. Influence of working frequency

In Fig. 5, influence of the working frequency on the cooling capacities and system COPs was investigated at the different cold-end temperatures with helium under 2.5 MPa mean pressure. The system COP was obtained by the cooling power divided by the electrical power input into the compressor at the corresponding cold-end temperature. It shows that the 15 Hz working frequency gave better performance of the refrigerator. In the experiments, a cooling power of about 200 W at -78°C has been achieved, which reached the temperature range of a conventional two-stage

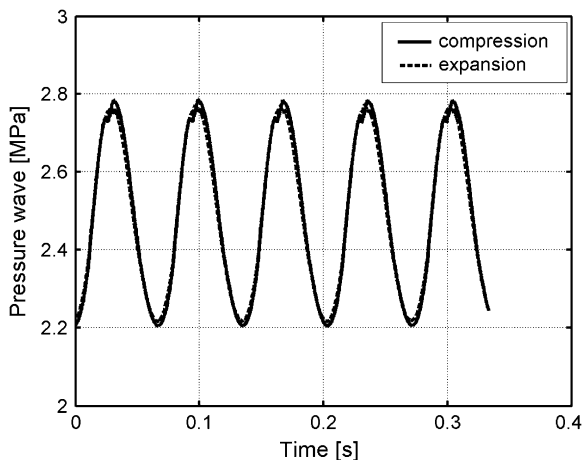


Fig. 3. Pressure waveforms under 15 Hz working frequency and 2.5 MPa mean pressure.

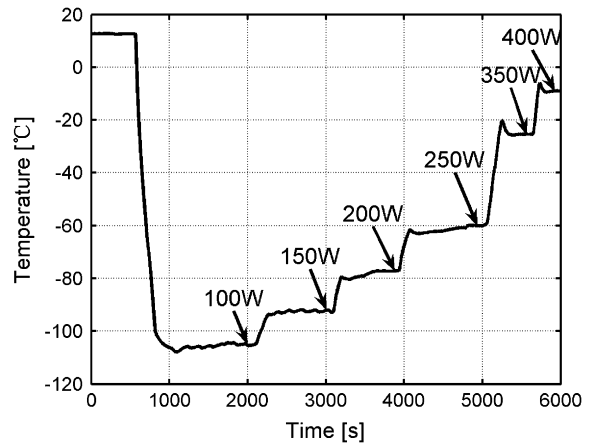


Fig. 4. Cool-down curve with cooling capacity measuring process under 15 Hz working frequency and 2.5 MPa mean pressure.

vapor-compression refrigerator. However, the cooling capacity and system COP went down as the working frequency increased.

4.4. Influence of mean pressure

Fig. 6 gives the experimental cooling capacities and system COPs under 20 Hz working frequency and different mean pressures as the function of the cold-end temperature. Obviously, we can obtain a better performance of the TBTSR at 2.0 MPa mean pressure with 20 Hz working frequency.

5. Discussions

Working duration and efficiency of the refrigeration systems are most concerned as well as environmental friendliness, cost and so on. As mentioned above, the TBTSR was

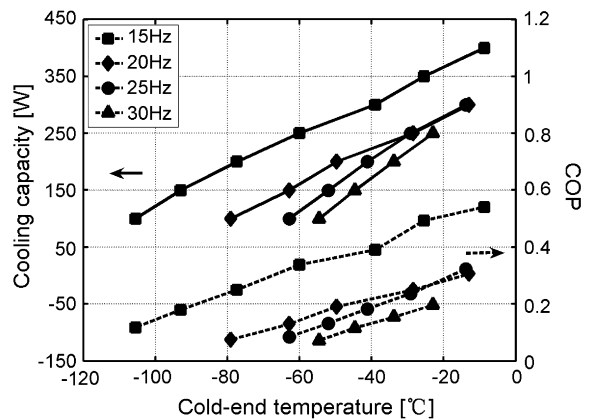


Fig. 5. Experimental cooling capacities and system COPs under 2.5 MPa mean pressure and different working frequencies. System COP was defined as the division between the cooling capacity and input electrical power of the compressor.

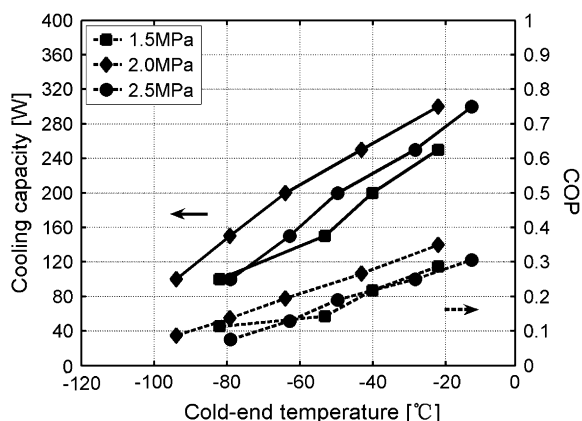


Fig. 6. Experimental cooling capacities and system COPs under 20 Hz working frequency and different mean pressures. System COP was defined as the division between the cooling capacity and input electrical power of the compressor.

completely environmentally friendly and the system cost was low because it used oil-lubricated compressors which were mature and well-developed. However, the system duration and the efficiency should be further discussed.

5.1. Working duration of the membrane

According to our experiments, the oil-lubricated compressor could be easily modified as a power source of the regenerative refrigerator by using membranes. However, the duration of the membranes would be a big challenge. In our design, the membrane was made from fluororubber and its deforming area was 1.7 bigger than the piston area and the duration of the membrane has been less than 100 h. To achieve long life duration of the membrane, three factors are important: (a) the membrane should be made from material uncorroded by oil, moreover it should be very soft; (b) the membrane area should be as large as possible to reduce deforming displacement, however thus would be limited by system configuration and would bring large void volume; (c) the mean pressure should be kept the same as system works, thus equilibrium position of the membrane would not shift away.

5.2. COP estimation of the TBTSR

In Section 4, we gave the COP of the whole system which was much lower than expected. The main reason could be that the compressor loss was predominant because the rating power of the compressor was 3.5 kW, which was greatly beyond the input electrical power in the experiments given in Fig. 7. To assess the performance of the cooler itself, we estimated the COP of the refrigerator by using cooling power to divide input acoustic work, which was obtained by subtracting input electrical power under vacuum condition from that under experimental conditions. Although this

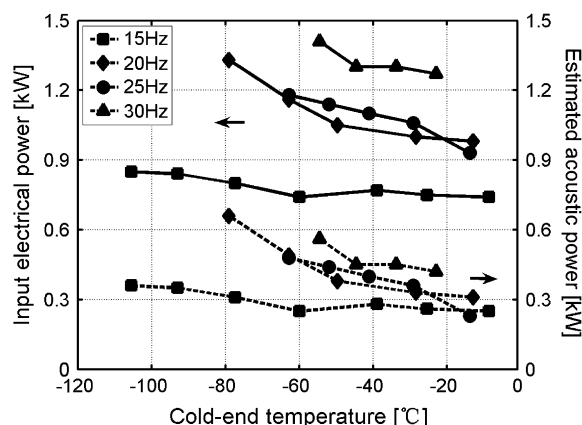


Fig. 7. Input electrical power and estimated acoustic power under different working frequencies. The estimated acoustic power was obtained by subtracting input electrical power under vacuum condition from that under experimental conditions.

method may underestimate the refrigerator COP because of the higher losses of the compressor under loaded condition than that under vacuum state, including gas flow losses and increased Joule losses due to increased electrical current. In the vacuum condition, the input electrical powers of the compressor were 490, 670, 700 and 850 W under the working frequency of 15, 20, 25 and 30 Hz, respectively. In Fig. 8, the TBTSR COPs were remarkably high especially under 15 Hz frequency since the impact of the compressor loss was approximately eliminated.

6. Conclusions

In this paper, Stirling refrigerator with a TBT driven by a modified commercial oil-lubricated compressor was proposed as a potentially alternative refrigeration method for

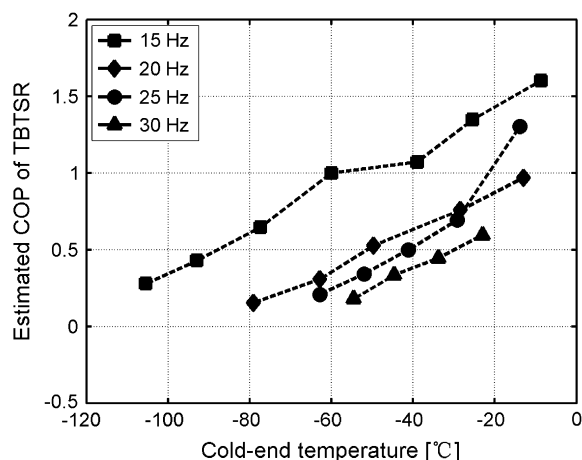


Fig. 8. Estimated COPs of the TBTSR under different working frequencies. Estimated COP was defined as the division between the cooling capacity and input acoustic work of the TBTSR.

domestic refrigeration purpose. The TBT was introduced to enhance the system reliability by moving the expansion piston to ambient location. Inside the modified oil-lubricated compressor, membranes were used to transfer acoustic work and keep the refrigerator working gas away from oil contamination. In the experiments, a better performance was achieved under 15 Hz working frequency and 2.5 MPa mean pressure and a cooling capacity of 200 W at -78°C is obtained. And the 2.0 MPa mean pressure with 20 Hz working frequency also gave a better result than other mean pressures. To evaluate the TBTSR performance more reasonably, we calculated the COP of TBTSR by means of eliminating the compressor loss, which used the subtraction between input electrical powers under experimental conditions and those under vacuum conditions as the input acoustic works of the TBTSR. The COPs of TBTSR were acceptable. This article has given a possible approach to use oil-lubricated compressor in regenerative refrigerator by applying membranes, whose material and configuration must be chosen and designed carefully.

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